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Thermo-Electro-Fluidic Simulation Study of Impact of Blower Motor Heat on Performance of Peltier Cooler for Protective Clothing

Kwon Joong Son 

Department of Mechanical and Design Engineering, Hongik University, Seoul 30016, Republic of Korea; kjson@hongik.ac.kr

Abstract: The necessity for portable cooling devices to prevent thermal-related diseases in workers wearing protective clothing in hot outdoor weather conditions, such as COVID-19 quarantine sites, is increasing. Coolers for such purposes require a compact design and low-power consumption characteristics to maximize wearability and operating time. Therefore, a thermoelectric device based on the Peltier effect has been widely used rather than a relatively bulky system based on a refrigeration cycle accompanying the phase change of a refrigerant. Despite a number of previous experimental and numerical studies on the Peltier cooling device, there remains much research to be conducted on the effect and removal of motor-related internal heat sources deteriorating the cooling performance. Specifically, this paper presents thermo-electro-fluidic simulations on the impact of heat from an air blower on the coefficient of performance of a Peltier cooler. In addition, a numerical study on the outcome of heat source removal is also evaluated and discussed to draw an improved design of the cooler in terms of cooling capacity and coefficient of performance. The simulation results predicted that the coefficient of performance could be raised by 10.6% due to the suppression of heat generation from a blower motor. Accordingly, the cooling capacity of the specific Peltier cooler investigated in this study was expected to be considerably improved by 80.6% from 4.68 W to 8.45 W through the design change.

Keywords: thermoelectric cooling; Peltier effect; multiphysics simulation; computational fluid dynamics; conjugate heat transfer



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1. Introduction

There exists a great risk of heat-related diseases when working in hot outdoor working conditions, such as at COVID-19 quarantine sites where wearing protective clothing is mandatory [1]. This is because protective clothing acts as a barrier to prevent the convection of inner air and body heat to the ambient environment. Therefore, a technical need for portable cooling devices suitable for protective clothing has emerged to enhance comfort while preventing the risk of heat-related illness, such as heat syncope, in hot weather. Furthermore, such portable cooling systems should be lightweight, compact, and low-power-consuming for long-lasting serviceability without interfering with quarantine tasks and charging during shift hours.

Depending on the energy source, protective clothing cooling devices can be classified into passive systems that use cold heat sources, such as ice packs, and active systems that use refrigeration cycles or thermoelectric cooling. Passive systems bear disadvantages such as difficulties regarding temperature control and air circulation. In addition, there is a risk of frostbite and poor comfort due to dew formation from cooling sources at very cold temperatures compared to body temperature [2,3]. The active system can control the temperature and flow rate of air during its operation. However, there may be the disadvantage of poor portability because high-density components such as batteries and motors must be equipped. In particular, refrigeration cycle-based coolers are unsuitable for

portable use in protective clothing since they must be equipped with pumps, compressors, and evaporators for refrigerant circulation, evaporation, and condensation [4,5]. On the other hand, thermoelectric cooling does not require complex refrigerant processes, thereby achieving miniaturization and weight reduction of the portable cooler [6,7].

Portability significantly differs depending on the cooling range and capacity, even in the same thermoelectric cooling method. A full-body cooling system has been developed to allow cooled air to circulate through the upper chest and back areas of the protective suits [6]. This cooling device also has a dehumidification function against moisture caused by sweat inside the protective clothing. Despite adopting such advanced technology, the system is heavy weighing 1.2 kg and equipped with two 80 mm-diameter circulation fans, making it difficult for workers to wear it for long periods of time. A half-body cooling system that supplies cold air only to the upper body has also been developed. The weight of the system, excluding the blower fans and batteries, is 450 g, showing a dramatic reduction in weight [8]. This paper introduces the design and performance of an even more lightweight Peltier cooler than the two devices mentioned above. The cooler, with a compact structure and weighing only 279 g, can blow cooled air into the protective clothing without the requirement of additional fans in the other parts of the clothing.

Experimental studies, system modeling studies, and numerical analysis studies have been conducted complementarily to design and evaluate the performance of portable Peltier coolers. Experimental studies aim to measure cooler prototypes' performance, identify technical issues, and draw design improvement plans where necessary. The testing apparatus for Peltier coolers has been designed and constructed to measure air flow rate and temperature, surface temperature distribution, fan rotation speed, and power consumption by using various sensors and data acquisition devices such as anemometers, thermocouples, thermographic cameras, electrical power sensors, motor encoders, and so forth [9–12]. For example, experimental studies have been conducted to determine the performance characteristics of the cooler by measuring the coefficient of performance (COP), a widely used figure of merit for thermoelectric cooling systems, against the blower fan speed or by measuring the electric consumption versus COP relation [11,12].

The system modeling approach can derive an initial design that fulfills the performance requirements from the experimental models or rules of thumb of various components constituting the Peltier cooler. For example, the performance curve of the blower fan [13], the heat pumping characteristic of the Peltier element [14], and the heat exchange model of the cooling fin [13] can be utilized in this method. Model-based design strategy is beneficial at the initial design and prototyping stages before the detailed development process through numerical and computational approaches. This can help component selection and system construction by predicting the device's cooling performance approximately [15].

When developing thermoelectric cooling devices, computer-aided engineering (CAE) methods have been widely exploited complementarily with experimental or system modeling approaches because computer simulations can verify the design rapidly and reliably before the prototyping and testing stages. The Peltier cooler operates inherently by coupling thermoelectric effects, fluid flow, and heat transfer. Therefore, a numerical method for it should have multiphysics characteristics [16,17]. Numerical modeling and analyses of Peltier coolers have been generally performed three-dimensionally rather than simplifying to lower dimensions due to the complex geometries of thermoelectric elements, heat exchanging fins, and fan blades. In terms of time dependence, both steady-state and transient analyses have been of great interest in this simulation field. This is because the prediction of both steady-state response and initial dynamic behavior is vital for the performance evaluation of a Peltier cooler, which takes a relatively long time of over 200 s to reach the steady state [18,19]. In the airflow analysis using computational fluid dynamics (CFD), turbulence models have been popularly used instead of laminar models because the associated Reynolds number is very high due to fast impeller speed and narrow cooling fin channels [8,9,13]. Comparing three widely used turbulent models, i.e., the standard $k - \epsilon$ model, the realizable $k - \epsilon$ model, and the shear stress transport (SST) $k - \omega$ model, the SST $k - \omega$

model showed the highest accuracy in a three-dimensional steady-state CFD analysis for a home refrigerator with an integrated Peltier cooling unit [19]. As a material constitutive relation, linear thermoelectric equations have been most widely used [20,21]. Their material coefficients are usually treated as constants, but temperature-dependent nonlinearity becomes non-negligible if the operating temperature difference is significant [22,23].

Three-dimensional steady-state CFD analysis of thermoelectric air cooling chamber with a liquid heat exchanger on the high-temperature side was carried out based on the realizable $k - \epsilon$ turbulent model [24]. Design verification of a portable Peltier air cooler integrated with the work jacket was performed with CFD simulations [8]. Transient CFD and thermoelectric analyses were conducted in a one-way coupling fashion [25]. While performing transient CFD analysis based on the $k - \omega$ turbulent model and transient thermoelectric analysis in order, the average wall temperatures on both TEC sides obtained from the CFD analysis were reflected in the thermoelectric model computation as boundary conditions. Although the blower fan geometry was included in the CFD analysis in the above-mentioned simulation studies, any heat transfer through the blower surface was neglected by treating it as an insulation boundary condition. To the best of the author's knowledge, computational research on the effects of fan motor heat on the performance degradation of thermoelectric coolers (TECs) for protective clothing has yet to be published, even though the amount of heat generation is considerable.

This paper introduces a numerical study on the effect of heat from a blower motor on the performance of a portable Peltier cooler for protective clothing. A three-dimensional, steady-state, coupled-field analysis based on an incompressible turbulent model and linear thermoelectric constitutive relations is performed for multiphysics phenomena in which turbulent airflow, thermoelectric cooling, and heat transfer through conduction and convection co-occur. The simulation results for the case incorporating the motor heat were verified by comparing them with the experimental results. In addition, a simulation without the internal heat source was carried out to quantify the improvement of cooling capacity and efficiency.

The rest of this paper is structured as follows. The consecutive Sections 2 and 3 introduce the design characteristics of the proposed Peltier cooler and its performance testing results, respectively. Section 4 presents the basic numerical models and analysis setups for the simulation of the Peltier cooler. Section 5 shows the simulation results with experimental observations and discusses the effect of an internal heat source on the overall cooling performance. Lastly, Section 6 presents the conclusions of this computational study.

2. Peltier Cooler Design

According to the opinions of the medical staff at COVID-19 quarantine sites, the main technical specifications of the cooler required for outdoor tasks in hot weather conditions can be summarized into four main items. First, the cooler for protective clothing should prevent potential heat diseases by supplying conditioned air at least 5 °C lower than the outside temperature in hot weather. Second, the cooler should operate continuously for at least two hours before the work shift without battery replacement or charging. Third, the cooler structure should be simple and compact, less than 10 cm in thickness, to facilitate its attachment and detachment to and from protective clothing as well as to avoid any interferences with tasks during operation. Fourth, the system's total weight, excluding the battery, should be less than 300 g so as to mitigate muscle fatigue in outdoor workers.

A portable Peltier cooler was designed to satisfy the four requirements mentioned above, and its prototype was built for immediate use in the field. Figure 1 shows the CAD image of the Peltier cooler (Cryo™ designed and manufactured by NK Innovation, Inc., Sejong, Korea) used in this computational study. The overall dimension of the cooler is 172 mm × 49 mm × 61 mm, and its weight is 279 g. The power consumption of the cooler is 30 W, which can be used for more than 2 h with a 11.1 V 8700 mAh rechargeable battery supplying 96.57 Wh at maximum. Structural materials should facilitate rapid prototyping and enable effective thermal insulation between cold and hot air ducts. Therefore, a 3-D

printable photoactive polymer resin (Formlabs™ White Resin) with low thermal conductivity and lightweight was used as the primary structural material. The thermoelectric element adopted for the cooling system is TEM TB-127-1.4-2.0 manufactured by Kyrotherm. Heat sinks were attached to both sides of the Peltier element, and the insulation wall between them separated cold and hot air ducts. Heat sinks are made of aluminum alloy with lightness and high thermal conductivity. Due to its high thermal conductivity and excellent adhesion characteristics, TSE3941 silicone adhesive from Momentive Performance Materials Inc. was used for heat sink attachment. In addition, each flow channel is equipped with a blower fan to ventilate air and induce forced convection around the heat sink.

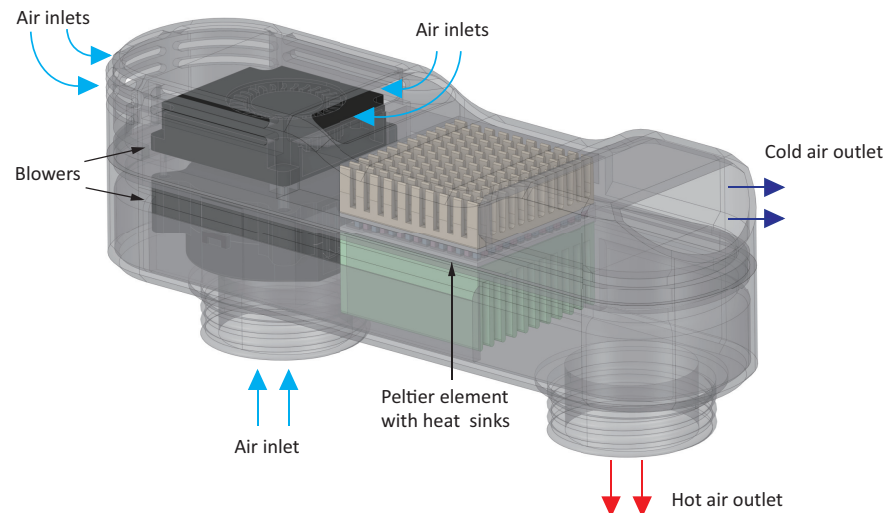


Figure 1. Design of portable Peltier cooler Cyro™ for protective clothing (courtesy of NK Innovation, Inc.).

Figure 2 shows the constructed prototype and an example of the protective clothing with the Peltier cooler installed. The Peltier cooler design in this study is similar to that of [9] in that the cooling and heating parts are separated for insulation but different in that the airflows are in the same direction, not in the opposite direction. As shown in Figure 2a, the device frame consists of three 3D-printed parts: the middle substrate and two duct covers with the air inlet and outlet. Figure 2b shows an example of the Peltier cooler installed inside a protective suit. The two holes are the hot air duct's inlet and outlet. The cooler and the battery pack are fixed by an inner chest harness, which accomplishes structural and installation simplicity.

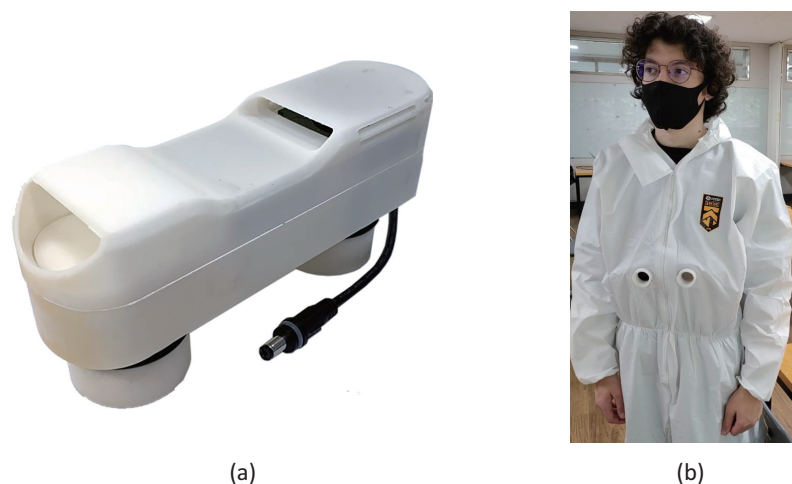


Figure 2. (a) Constructed cooler prototype, (b) protective clothing with Peltier cooler attached to inner chest harness (courtesy of NK Innovation, Inc.).

The blowers in the air ducts have an integrated structure of the motor and the impeller. They inhale stagnant outer air in the direction of the impeller's axis of rotation and blow it toward the heat sink for heat exchange. In this process, the heat from the motor is unwantedly transferred to the air, causing a preheating effect before reaching the heat sink. As a result, the blower improves heat-exchanging efficiency through forced convection at the low-temperature heat sink while also lowering cooling performance by the air-preheating effect. As the motor output boosts to enhance blowing capacity, the degradation in cooling efficiency due to preheating becomes more significant. Similarly, the preheating effect from the hot side blower also deteriorates the heat dissipation capability at the high-temperature heat sink. Therefore, this paper analyzes the impact of motor heat on the thermoelectric cooling performance using multiphysics simulations and predicts the potential performance improvement by removing such internal heat sources.

3. Performance Test Results

An experimental study was conducted to evaluate the performance of the Peltier cooler prototype and obtain experimental data to validate the simulation results. Figure 3 shows the experimental apparatus for testing the cooler's performance. Figure 3a,b present an experimental setup for the cold and the other side duct, respectively. The apparatus comprises a structural frame, various sensors, and a data collection board with a flat-panel display. The main frame for fixing the cooling device and the sensing components is the assembled structure of aluminum extrusion profiles. A pair of thermal anemometers measured the velocity and temperature of the air in both the inlet and outlet. Although not shown in Figure 3, the non-contact thermal camera module (Letpton 3.5 manufactured by Teledyne FLIR LLC, Wilsonville, OR, USA) and a portable thermal camera (GTC-600C supplied by Robert Bosch GmbH, Gerlingen, Germany) monitored the surface temperature of the subject device. The opaque plastic cover was partially cut and replaced with a transparent film, as shown in Figure 3a, to capture the surface temperature of the heat sink and the blower inside the cold air duct with thermal cameras. Power digital wattmeters (Model SEN0291 by Zhiwei Robotics Corp., Shanghai, China) were used to measure power consumption by sensing the shunt voltage and the electric current for electric components such as the thermoelectric element and the blower motors. A single-board computer (Raspberry Pi 4 Model B made by Raspberry Pi Foundation, Cambridge, UK) processed the wattmeter data to be displayed on the monitoring panel.

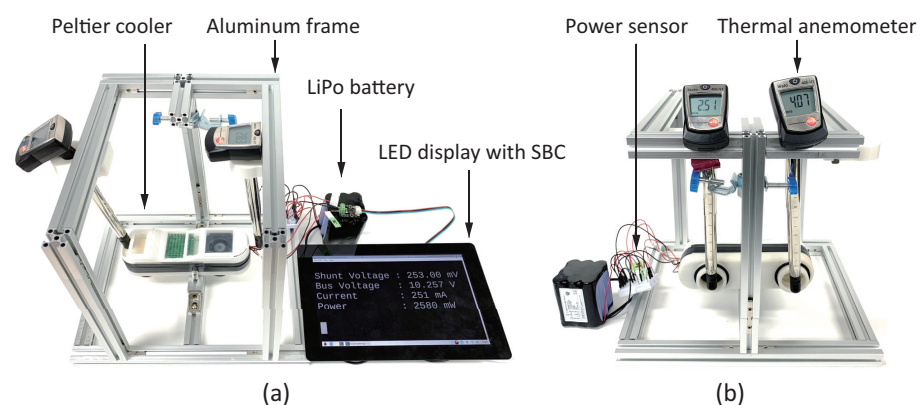


Figure 3. Experimental apparatus: configuration for performance test (a) on the cold duct side and (b) on the hot duct side.

Coefficient of performance (COP) is a widely used performance figure for Peltier coolers [11,12,26]. COP is defined by the ratio of cooling capacity Q_c to total power consumption P_t and can be expressed as

$$\text{COP} = \frac{Q_c}{P_t} = \frac{Q_c}{P_p + P_b} \quad (1)$$

where P_p and P_b are the power consumption of the Peltier element and the blowers, respectively.

For the Peltier cooler of the design shown in Figure 1, the cooling capacity Q_c represents the net thermal energy reduction rate in the cold air duct [24], so it can be expressed as

$$Q_c = (\dot{m}C_pT)_{\text{inlet}} - (\dot{m}C_pT)_{\text{outlet}} \quad (2)$$

where $\dot{m}$ is the mass flow rate, C_p is the heat capacity at constant pressure, and T is the absolute temperature.

Figure 4 shows the temperature distribution plots for the portable Peltier cooler in operation obtained by the infrared thermal imaging camera. The minimum temperature on the upper surface of the heat sink in the cold air duct was measured as 20.8 °C. Meanwhile, the maximum temperature on the blower surface was 44.7 °C and 47.8 °C in the cold and hot air ducts, respectively. These measured surface temperatures were used as the essential boundary conditions imposed on the blower surfaces for heat transfer analysis.

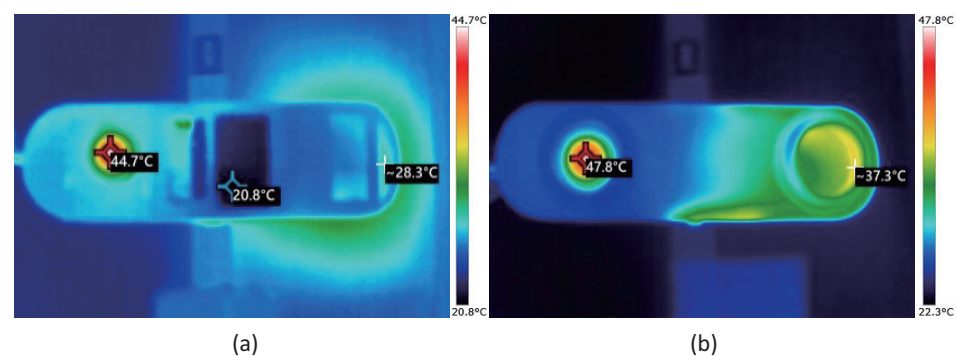


Figure 4. Color-mapped infrared thermal images (a) on the cold duct side and (b) on the hot duct side.

Table 1 shows the experimental results and the calculated performance indices. Each measured value in this table is the mean of three data points in a steady state with a low data fluctuation. The COP of the cooler, including the internal heat source, was evaluated as 10%. The air temperature could be reduced by 1.9 °C at most, falling short of the design target of 5 °C.

Table 1. Experimental results and calculated performance values.

Item	Value
Ambient temperature	26.0 °C
Power consumption (Peltier element)	22.67 W
Power consumption (cold duct blower)	2.91 W
Power consumption (hot duct blower)	3.77 W
Cold duct outlet velocity	4.06 m/s
Cold duct outlet temperature	24.10 °C
Hot duct duct outlet velocity	4.05 m/s
Hot duct outlet temperature	36.06 °C
Cold duct heat sink base temperature	20.8 °C
Cold duct blower surface temperature	44.7 °C
Hot duct blower surface temperature	47.8 °C
Cooler capacity ¹	5.11 W
COP ²	17.4%

¹ The cooling capacity Q_c was obtained from Equation (2). ² The coefficient of performance was calculated from Equation (1).

4. Computational Model and Analysis Setup

This section presents the fundamental theories and simulation setup for carrying out a three-dimensional steady-state thermo-electro-fluidic analysis of the portable Peltier cooler introduced previously.

4.1. CFD and Thermoelectricity Model

The governing equations for the Peltier cooler analysis, which simultaneously incorporates the fluid flow, electrical current, and temperature fields, comprise the mass continuity equation, the electrical continuity equation, the momentum conservation equation, and the energy conservation law. Without describing those well-known fundamental governing equations, this section only introduces other basic theories for understanding the coupled-field simulation approach.

The specific turbulence model used in this research is the shear stress transport (SST) $k - \omega$ model, which combines the advantages of the $k - \epsilon$ model and the $k - \omega$ model [27,28].

The SST $k - \omega$ model consisting of two equations for the specific turbulent kinetic energy k and its dissipation rate ω can be expressed in the steady-state form [29] as

$$\frac{\partial}{\partial x_j}(\rho k u_j) = \tau_{ij} \frac{\partial u_i}{\partial x_j} - \beta^* \rho \omega k + \frac{\partial}{\partial x_j} \left[(\mu + \sigma_k \mu_t) \frac{\partial k}{\partial x_j} \right] \quad (3)$$

$$\frac{\partial}{\partial x_j}(\rho \omega u_j) = \frac{\gamma}{\nu_t} \tau_{ij} \frac{\partial u_i}{\partial x_j} - \beta \rho \omega^2 + \frac{\partial}{\partial x_j} \left[(\mu + \sigma_{\omega 1} \mu_t) \frac{\partial \omega}{\partial x_j} \right] + 2(1 - F_1) \rho \sigma_{\omega 2} \frac{1}{\omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} \quad (4)$$

where ρ is the mass density, u is the fluid velocity component, τ is the Reynolds stress tensor, μ is the dynamic viscosity, μ_t is the turbulent viscosity, and ν_t is the eddy viscosity. In Equations (3) and (4), β^* , β , σ_k , $\sigma_{\omega 1}$, $\sigma_{\omega 2}$ and γ are constants computed based on two sets of model constants, i.e., one for the Wilcox $k - \omega$ model and the standard $k - \epsilon$ model using a lever rule with the weight factor F_1 . For those interested in further details on the derivation of the two-equation SST $k - \omega$ turbulent model and the associated model parameters, it is highly recommended to see the reference article [27].

The Prandtl number, defined as the kinematic viscosity to thermal diffusivity ratio, can compute heat transfer through interfaces between air and heat sinks. In this study, the Kays–Crawford turbulent Prandtl number model in the following form [30] was chosen for the conjugate heat transfer (CHT) analysis of the Peltier cooler.

$$Pr_t = \left[\frac{1}{2Pr_{t\infty}} + \frac{0.3}{\sqrt{Pr_{t\infty}}} \frac{\mu_t}{\rho D} - \left(\frac{0.3\mu_t}{\rho D} \right) \left(1 - e^{-\frac{\rho D}{0.3\mu_t \sqrt{Pr_{t\infty}}}} \right) \right]^{-1} \quad (5)$$

In the above Equation (5), Pr_t is the turbulent Prandtl number, $Pr_{t\infty}$ is its far-wall value, and D is the diffusion coefficient. Readers interested in further details of the Kays–Crawford model used in this simulation are suggested to review the paper [31].

The heat flux and temperature associated with Peltier devices are mostly computed analytically by the simple heat flux model or numerically by the finite element method (FEM) with thermoelectric constitutive relation [21,26,32]. Thermoelectric constitutive equations for the electric current density $\mathbf{J}$ and the heat flux $\mathbf{q}$ can be written as a function of the electric field intensity $\mathbf{E}$ and the absolute temperature T in a vector form [33], respectively, as

$$\mathbf{J} = \sigma(\mathbf{E} - \alpha \nabla T) \quad (6)$$

$$\mathbf{q} = \pi \mathbf{E} - \kappa \nabla T \quad (7)$$

where σ is the electric conductivity, α is the Seebeck coefficient, ∇ is the spacial gradient operator, $\pi = \alpha T$ is the Peltier coefficient, and κ is the thermal conductivity. For problems with a steep temperature gradient in the thermoelectric element, the temperature dependency of the material constants in Equations (6) and (7) should be considered [22].

4.2. Simulation Setup

As the first step in the pre-processing, simulation domains subject to electro-thermo-fluidic analysis were selectively extracted from the CAD model illustrated in Figure 1.

Among shape details, small-sized features were eliminated, such as small rounds or steps, since they may cause poor computational efficiency due to unnecessary element refinement without significantly affecting computational accuracy. Subsequently, the connectivity between CAD components was checked and modified where necessary to extract the three-dimensional volumes corresponding to the fluid domains so that each cavity space had a well-defined continuous and closed surface. In total, two fluid domains corresponding to cold and hot air ducts were created from the cavity of the CAD assembly model after small feature removal and continuity check. After the fluid volume creation, solid parts, such as the plastic covers and fan blades, not included in the computational analysis were removed. Thin adhesive layers between the thermoelectric part and the heat sinks were modeled as shell elements, not three-dimensional elements. The thickness was set to 500 μm to calculate the adhesive layer's thermal resistance and temperature gradient using the shell element.

Figure 5 depicts the finalized computational domain through the above-mentioned process together with the mesh elements created by COMSOL Multiphysics software. To clearly show the internal features of the computational domain, the geometry was treated translucently, and the meshed elements were marked with lines only on the outer surface. The fluid regions consist of 14,575,305 elements. In addition, the solid volumes corresponding to the thermoelectric element and heat sinks were discretized into 865,592 elements. For the silicon adhesive thin film, 6463 surface elements were allocated. The entire computational domain has 3,533,910 associated nodes.

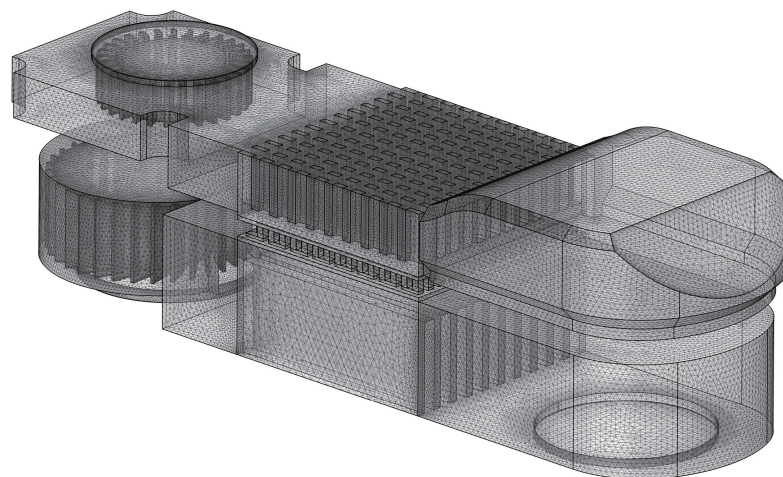


Figure 5. Simulation domains with meshed elements.

The thermoelectric element has a composite structure made of bismuth telluride (Bi_2Te_3), copper, and alumina, whose fundamental properties are shown in Table 2. Unlike other materials, in the case of bismuth telluride, which has a relatively large temperature gradient due to the Peltier effect, the temperature-dependent property was adopted as in Table 3. This is the data look-up table contained in the material database of COMSOL Multiphysics to account for such nonlinear material properties. Seebeck coefficient, electric conductivity, and thermal conductivity can be computed using a cubic spline interpolation scheme on the five data listed in Table 3. The physical properties of air for fluid domains, aluminum for heat sinks, and silicone for an adhesive applied to the heat sink and thermoelectric element interfaces are shown in Table 4.

The fluid analysis domain consists of two parts, i.e., cold and hot air ducts, according to the cooler's structure. Each duct has a blower fan for forced convection, and the fan motor operates at a high speed of about 9000 rpm, which generates rotational turbulence. The rotational flow simulation itself, which incurs a very high computing cost, is beyond the interest and scope of this study. Instead, for computational efficiency, a flow field under the same volume flow rate without consideration of the fan motion was computed to account for the forced convection effect around the heat sink. In this flow analysis process, the inlet

velocity condition was derived from the measured volume flow rate at each duct, and the outlet condition was set as the atmospheric pressure. The remaining boundaries, except the inlet and outlet surfaces, were all treated as a no-slip wall. The fan motor heat was assumed to outflux through the core cylindrical surface, excluding blades. Although the blower input power can be measured with the experimental apparatus shown in Figure 3, the heat flux cannot be estimated through accurate motor efficiency measurement as in [34]. The internal heat source effect was reflected in the simulation by assigning the measured surface temperature of the blower to the relevant boundaries instead of imposing the motor's energy dissipation rate. All boundary conditions applied to the fluid domain described above are summarized in Table 5.

Table 2. Properties of constituting materials thermoelectric element (from material database of COMSOL Multiphysics).

Material	Density (kg/m ³)	Heat Capacity (J/kg·K)	Thermal Conductivity (W/m·K)	Electrical Conductivity (S/m)	Seebeck Coefficient (V/K)
Bi ₂ Te ₃	7700	154	$\kappa(T)^1$	$\sigma(T)^1$	$\alpha(T)^1$
Copper	8960	385	400	5.998×10^7	-
Alumina	3900	900	27	-	-

¹ Refer to Table 3 for temperature-dependent material data.

Table 3. Temperature-dependent properties of Bi₂Te₃ (from material database of COMSOL Multiphysics).

Temperature (K)	$\kappa(T)$ (W/m·K)	$\sigma(T)$ ($\times 10^5$ S/m)	$\alpha(T)$ ($\times 10^{-4}$ V/K)
200	2.4	1.4286	1.68
250	1.9	1.1111	1.92
300	1.6	0.86957	2.10
350	1.6	0.71429	2.25
400	1.75	0.58824	2.37

Table 4. Material properties of air and structural components (from material database of COMSOL Multiphysics).

Material	Density (kg/m ³)	Heat Capacity (J/kg·K)	Thermal Conductivity (W/m·K)	Viscosity (m ² /s)
Air	1.184	1007	0.02551	1.563×10^{-5}
Aluminium	2700	900	238	-
Silicone	1650	1460	0.83	-

This study adopted a partial coupling approach for selected energy domains in some computation stages instead of a full coupling scheme. The partially coupled solver is highly advantageous regarding computing time, although the calculation accuracy is slightly less than that of the fully coupled solver in a tolerable range [23]. Specifically, before thermoelectric and conjugate heat transfer analyses, turbulent flow and electric current analyses were carried out to obtain the velocity field in the fluid domain and the current density field in the electric domain, respectively. Then, the computed field data were transferred to the subsequent computing steps in a one-way coupling fashion. Based on the shared flow and electric field data, in the remaining steps, the temperature field was calculated in a fully coupled manner which simultaneously incorporates the temperature-dependent Peltier effect, Joule heating, and heat transfer through conduction and convection.

Table 5. Boundary conditions for simulation study.

Boundary Condition	Value	Remarks
Cold duct inlet velocity	3.94 m/s	Derived from measured ¹
Cold duct inlet temperature	26.03 °C	Measured
Cold duct blower temperature	44.7 °C	Measured
Hot duct inlet velocity	4.05 m/s	Derived from measured ¹
Hot duct inlet temperature	26.03 °C	Measured
Hot duct blower temperature	47.8 °C	Measured
Input voltage	10.38 V	Measured

¹ The inlet velocity was calculated from the continuity equation with incompressibility assumption.

5. Simulation Results and Discussion

Numerical simulations to evaluate and improve the thermoelectric cooling performance of the portable Peltier cooler shown in Figure 2 were conducted in a coupled fashion using the commercial software package COMSOL Multiphysics with its extension modules, including CFD Module, AC/DC Module, and Heat Transfer Module.

According to the presence of heat sources in the air ducts due to the fan motor operation, numerical simulation was conducted in two cases: one incorporating the internal heat and the other suppressing the motor heat. Each simulation was performed through three stages: turbulent CFD analysis, electric field analysis, and conjugate heat transfer analysis accompanying thermoelectric cooling and Joule heating effects. As a result, such a multiphysics simulation yielded the flow and pressure fields of the air, the potential and current density fields of the electric parts, and temperature fields throughout the entire computational domain.

Figure 6 shows the fluid velocity fields visualized as vector arrows in the cold and hot air ducts. The cold duct has a maximum speed of 27.32 m/s in the entire domain and an average normal velocity of 6.28 m/s through the middle cross-section of the heat sink. Meanwhile, the highest fluid speed in the hot air duct and the average cross-sectional velocity in the heat sink was calculated as 19.63 m/s and 7.60 m/s, respectively.

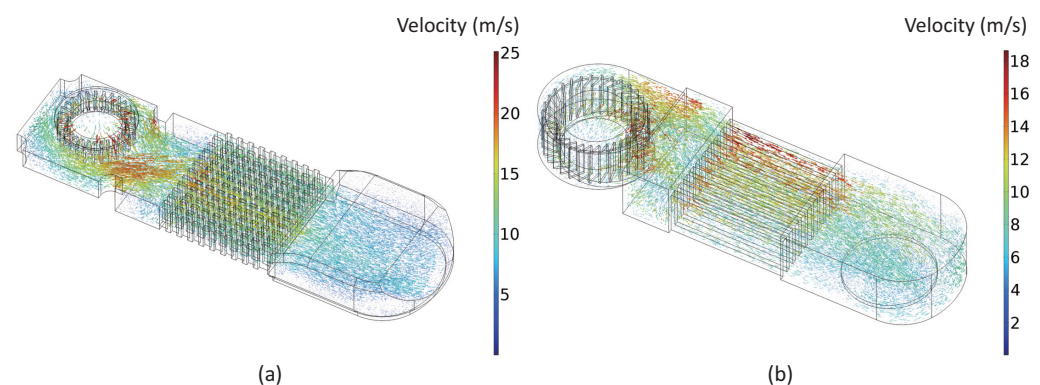
**Figure 6.** Simulation results: air velocity fields (a) in cold air duct and (b) hot air duct.

Figure 7 shows the simulation results in the electric domains, i.e., the copper wiring layers and semiconductor segments. Figure 7a displays the computed electric potential distribution by applying 10.38 V to the input port. Since the junctions of p-type and n-type semiconductors interconnected by copper conductors occur repeatedly in space, a uniform voltage drop appears across each junction. Figure 7b illustrates the electrical field intensity distribution with a color map and the current density with vector arrows at 30,000 different Gauss quadrature points. The length of each arrow was set in proportion to the vector magnitude of the corresponding current density. Due to the inherent characteristics, highly conducting copper plates possess nearly zero electric field intensity but relatively high current density. On the other hand, semiconducting bismuth telluride blocks have a higher electric field intensity of 23.2 V/m and a relatively lower current density than conductors.

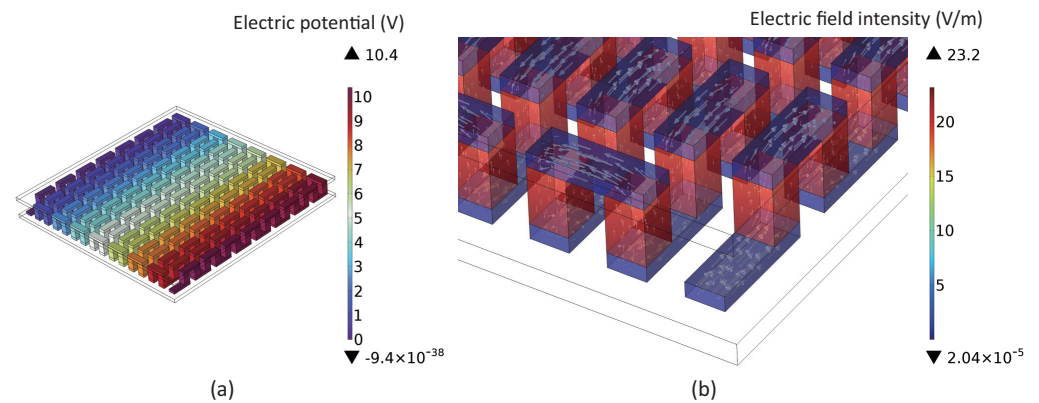


Figure 7. Simulation results: (a) electric potential distribution and (b) electric field intensity distribution with current density vectors.

Figure 8 illustrates the post-processed temperature distribution plots for the two simulation cases. The outer surfaces were set to be translucent to reveal the internal wall temperature features clearly.

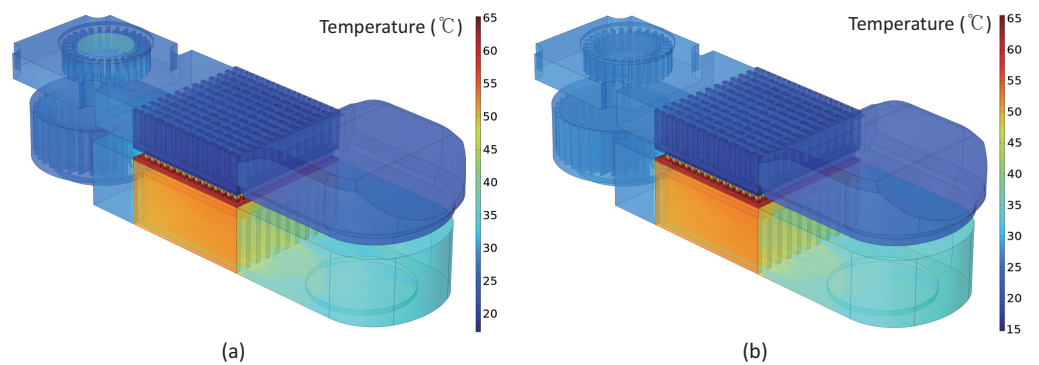


Figure 8. Temperature distribution plots for the Peltier cooler (a) including the internal heat sources and (b) with the heat generation suppressed.

To verify the reliability of these simulation results, Figure 9 compares the numerically computed temperature distribution with an experimentally obtained thermographic image. It should be noted that the thermal image in Figure 9b displays the external wall temperature, except for the internal wall temperature of selected portions replaced with the transparent film that can be observed in Figure 3a. On the other hand, Figure 9a shows the temperature distribution on the fluid boundaries with top surfaces virtually hidden. In other words, in Figure 9, it is only valid to compare the temperature distribution in two boundaries, i.e., the blower's circular top plane and the heat sink surface. Direct comparison in temperature of the other surfaces is not compatible. The minimum temperature on the heat sink surface was 20.8 °C in the experiment, while it was computed as 4.8% lower at 19.7 °C in the simulation. Therefore, it was confirmed that the simulation results predict the experimental results precisely within a 5% error.

Figure 10 depicts a bar chart comparing the average temperatures computed at three locations of interest from the temperature fields numerically obtained for the two simulation cases shown in Figure 8. The minimum heat sink surface temperature, the cold duct mean outlet temperature, and the hot duct average outlet temperature were 19.7 °C, 24.26 °C, and 30.04 °C, respectively, in the simulation accounting for the blower heat generation. In the other simulation with no account for the heat generation effect, meanwhile, the minimum heat sink surface temperature, the cold duct mean outlet temperature, and the hot duct average outlet temperature were 17.6 °C, 22.86 °C, and 35.4 °C, respectively, which were all lower than the simulation results considering blower heat.

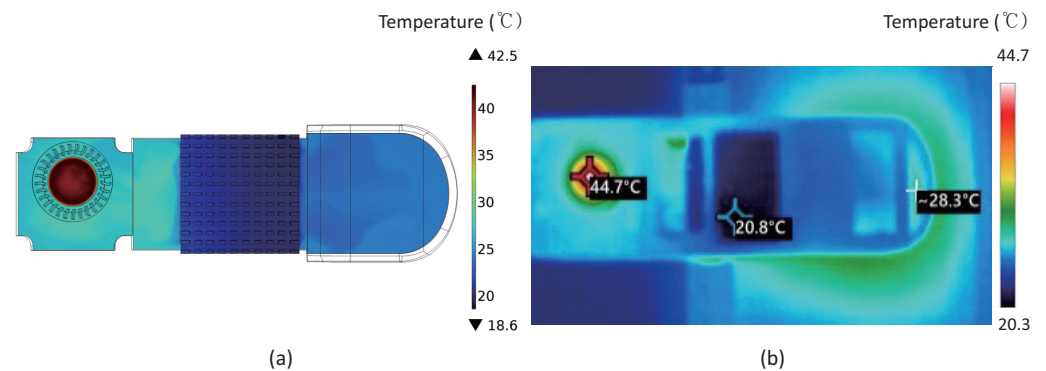


Figure 9. Temperature distribution comparison of (a) post-processed simulation image versus (b) experimentally captured thermal image.

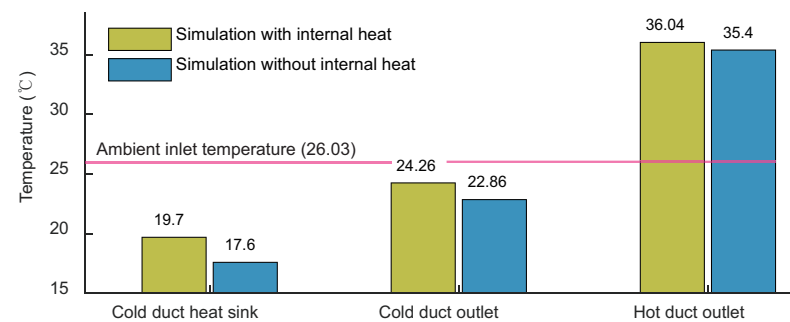


Figure 10. Temperature comparison chart at three locations according to the presence of internal heat source in simulation.

Figure 11 depicts the color-contoured temperature distribution only in the electric domain and the thermal energy transfer represented by vector arrows at 5000 different Gauss quadrature points. The length of each arrow was set in proportion to the vector magnitude of the heat transfer. The simulation results illustrate that the Peltier effect pumps the thermal energy from the top low-temperature region to the bottom high-temperature region so that the air can refrigerate in the upper chamber. In addition, this numerical analysis confirmed that the Peltier effect becomes more active due to the removal of the motor heat, which leads to further lower temperatures in the low-temperature side of the thermoelectric cooling module.

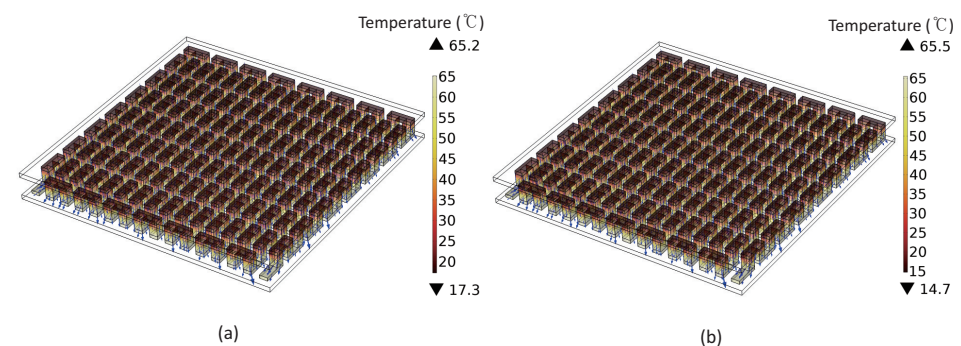


Figure 11. Temperature distribution plot in the electric domain due to the Peltier effect and corresponding heat flux vectors represented by arrows for two simulation cases (a) with account for internal heat sources and (b) without motor heat generation.

Comprehensively summarizing the above-described simulation results, Table 6 lists the experimental results and the calculated performance figures of merit. In this table, the total power consumption P_t is obtained as in the denominator of Equation (1) by

adding Peltier power consumption P_p obtained from the simulation and blower power consumption P_b measured from the experiment.

Table 6. Performance values derived from simulation results.

Variable	Internal Heat Included	Internal Heat Excluded
Current	2.8128 A	2.8128 A
Cold duct outlet temperature	24.26 °C	22.86 °C
Heat sink base temperature	19.7 °C	17.6 °C
Hot duct outlet temperature	36.04 °C	35.40 °C
Peltier power consumption	29.20 W	29.20 W
Total power consumption ¹	35.88 W	35.88 W
Cooling capacity ²	4.68 W	8.45 W
COP ³	13.0%	23.6%

¹ It was calculated by adding the simulated Peltier power consumption and measured blower power data. ² The cooling capacity Q_c was obtained from Equation (2). ³ The coefficient of performance was calculated from Equation (1).

Figure 12 comparatively shows the cooling capacity and COP derived from the performance test and two simulation cases as a bar chart. Even considering that the simulation results slightly underestimate the experimental results, it can be concluded from these thermo-electro-fluidic simulations that removing the internal heat source can significantly improve the cooling capacity by 80.6% from 4.68 W to 8.45 W, and accordingly, the COP by 10.6% from 13.0% to 23.6%.

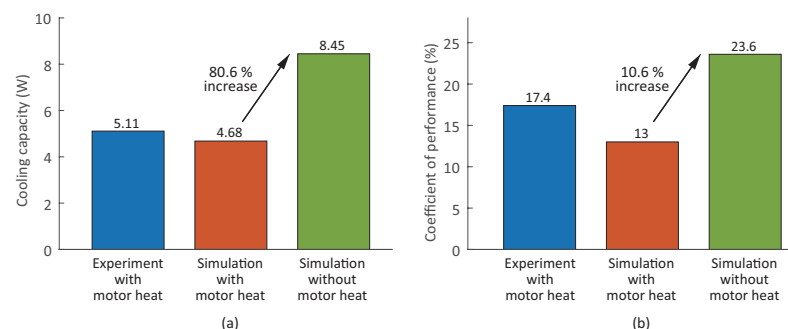


Figure 12. Bar charts for Performance figures of merit: (a) cooling capacity from Equation (2) and (b) coefficient of performance from Equation (1).

Internal heat sources can be technically removed in a way that the electric motors located outside the air ducts transmit the rotational power inward through the shafts. Cryo™, the Peltier cooling device for protective clothing shown in Figure 2a, satisfies three of the four technical requirements specified in Section 2, i.e., operating time, size, and weight, except for temperature reduction by 5 °C. The next version of Cryo™, with improved cooling performance through design optimization and heat removal inside the air duct, is currently under development through industry–academia cooperation.

6. Conclusions

The development of Peltier coolers for protective clothing to prevent the risk of heat-related diseases in hot working environments has been actively conducted. In particular, there has been great interest in energy-efficient cooling technology that can increase system portability and duration time by reducing weight and power consumption. However, little research effort has been exerted to investigate the effect of blower motor heat on overall cooling efficiency. Therefore, this paper aims to numerically evaluate the effect of heat from a blower motor on the performance of a portable Peltier cooler for protective clothing. Due to the multi-physical characteristics of the thermoelectric coolers, in this study we conducted the thermo-electro-fluidic analysis in a coupled fashion. Such a coupled-field

analysis can computationally evaluate the Peltier cooler's performance indices, such as cooling capacity and coefficient of performance, by incorporating turbulent flow analysis, electric current density analysis, Peltier effect analysis, Joule heating analysis, and conjugate heat transfer analysis. The reliability of the simulation results was validated through comparison with the performance test results for the Peltier cooler prototype. The simulation error against the measured data was 4.8% for the minimum heat sink surface temperature in the cold air duct, guaranteeing simulation accuracy at a tolerable level. The numerical analysis predicted that the cooling capacity and coefficient of performance could be improved by 50.6% and 13.0%, respectively, by installing the heat-generating blower motor outside the air ducts. The research findings of this study are differentiated in that they quantitatively analyzed the effect of heat generation in blower motors on Peltier cooler performance, which has not been covered in previous studies. The design and optimization of a novel Peltier cooler resolving the motor heating issue could present a potential topic of future research.

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Nomenclature

C_p	heat capacity at constant pressure, J/kg·K
D	diffusion coefficient, m ² /s
E	electric field intensity, V/m
F_1	$k - \omega$ model parameter weighting factor
J	current density, A/m ²
k	specific turbulent kinetic energy, J/kg
$\dot{m}$	mass flow rate, kg/s
P_b	power consumption of blowers, W
P_p	power consumption of Peltier element, W
P_t	total power consumption, W
Pr_t	turbulent Prandtl number
$Pr_{t\infty}$	far-wall turbulent Prandtl number
q	heat flux, W/m ²
Q_c	cooling capacity, W
T	absolute temperature, K
u	fluid velocity component, m/s

Creek symbols

α	Seebeck coefficient, V/m
β	$k - \omega$ model parameter
β^*	$k - \omega$ model parameter
γ	$k - \omega$ model parameter
ϵ	specific turbulent energy dissipation rate in $k - \epsilon$ model, J/kg·s
κ	thermal conductivity, W/m·K
μ	dynamic viscosity, Pa·s
μ_t	turbulent viscosity, Pa·s
ν_t	eddy viscosity, m ² /s
π	Peltier coefficient, J/C
ρ	mass density, kg/m ³

σ	electric conductivity, S/m
σ_k	$k - \omega$ model parameter
$\sigma_{\omega 1}$	$k - \omega$ model parameter
$\sigma_{\omega 2}$	$k - \omega$ model parameter
τ	Reynolds stress tensor, Pa
ω	specific turbulent kinetic energy dissipation rate, J/kg·s

Abbreviations

CAD	computer-aided design
CAE	computer-aided engineering
CFD	computational fluid dynamics
CHT	conjugate heat transfer
COP	coefficient of performance
FEM	finite element method
rpm	revolutions per minute
SST	shear stress transport
TEC	thermoelectric cooler

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